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DRAG-COMPENSATED, PRECISION-POWERED HINGE SYSTEM

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SUMMARY

The design of a high-precision powered hinge is complicated by the unavoidable presence of parasitic drag torque resulting mainly from friction and transfer of power, signals, and fluids across the hinge. Regardless of the type of drive system selected, it is impossible to completely eliminate all parasitic drag. However, the mechanism described here comes very close to providing a drag-free system. All sources of parasitic drag torque are collected on a shaft which is powered by an electric motor independent of the main hinge drive. Under control of a sensor, the electric motor applies a compensating torque equal to that of the parasitic drag torque, allowing the main hinge drive to operate in a practically drag-free environment with very high positioning precision.

INTRODUCTION

In the design of robotic arms, precision pointing systems and other mechanisms which require very accurate angular positioning, it is necessary to find methods for minimizing or eliminating the parasitic drag torque. The presence of parasitic drag torque introduces step functions into the torque-vs.-displacement curve, the nonlinearity of which is further aggravated by the effect of static friction. The parasitic drag is difficult to predict. It is known to vary with angular position, velocity, load, temperature, and other variables, especially if wire bundles and flex hoses are routed around the hinge. If the hinge drive mechanism design includes a gearbox gear backlash introduces deadbands in which positioning cannot be controlled.

The powered hinge mechanism presented in this paper eliminates gearing and its associated problems by using direct-drive motors, and reduces parasitic drag torque to that of one lightly loaded ball bearing.

POWERED HINGES - GENERALITIES

Powered hinges can be classed into two broad categories:

1. Deployables (nonretractable)
2. Remotely controlled continuously adjustable

The deployable systems of category 1 are usually spring-driven with deployment rates controlled by adjustable dampers. Precision positioning in the deployed position is provided by the lockup mechanism. This type of powered hinge is of no interest in the following discussion.

The remotely controlled systems of category 2 cover an array of devices adapted to various degrees of positioning precision, ranging from simple powered door actuators to robotic arm hinges and high-precision pointing systems. If high-precision pointing is not a requirement, relatively simple

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devices can be designed using gear-train reduction drives. Such mechanisms can sometimes be tolerant of significant parasitic drag torque so that transfer of power and electric signals can be performed externally by means of wire bundles or slip rings and fluids via flexhoses. However, the design of a hinge system with the high pointing precision of a fraction of an arc sec. presents difficult problems which cannot be resolved by simple conventional mechanisms. To meet such requirements, it is necessary to use more complex electromechanisms.

GENERAL REQUIREMENTS

The high-precision powered hinge system discussed in this paper was designed to meet the following specifications:

1. Maneuver an 8000-lb payload at the end of a 75-in. arm in a vacuum/zero-g environment.
2. Achieve fine pointing with a 0.50 arc sec. resolution and within ±2 arc sec. of the specified value.
3. Ensure that the jitter does not exceed:

Freq., Hz.	Max. jitter, arc sec.
0 to 5	0.60
5 to 20	0.50
20	0.20

4. Provide for electrical power transfer across the hinge (two lines).
5. Provide for signal transfer across the hinge (40 channels).
6. Provide for fluid line transfer across the hinge (four lines) or into the hinge systems, if cooling is required.

The pointing precision requirements are not too meaningful when expressed in terms of arc sec. In order to give a better appreciation of their severity, they are expressed below in inches at 1 mile from the hinge point.

pointing:	±2 arc sec.	=	± 0.62 in. at 1 mile
resolution:	0.50 arc sec.	=	0.16 in. at 1 mile
jitter:	0.60 arc sec.	=	0.19 in. at 1 mile
	0.50 arc sec.	=	0.16 in. at 1 mile
	0.20 arc sec.	=	0.06 in. at 1 mile

BASIC CONCEPT

To provide a mechanism which will perform with the high precision consistent with the above specifications, it is necessary to neutralize the parasitic drag torque such that the drive motor effectively operates a drag-free system. It is also necessary to use DC torque motors in direct drive to eliminate difficulties inherent with reduction drives and/or stepper motors.

The neutralization of the parasitic torque drag is provided by means of a drag torque eliminator system (DTES). The DTES senses the presence of drag torque and applies additional power independently from the main drive motor in such a manner that the drag torque is balanced out without disturbing the main drive.

The main drive is provided by a DC torque motor controlled by an electronic feedback control system which ensures the appropriate pointing precision.

The transfer of electric power is performed by means of rotary transforme using AC current. Similarly, the transfer of all electrical signals is provided via rotary capacitive couplers. Advantage is therefore taken of the property of both rotary transformers and capacitive couplers which perform their functions through air (or vacuum) gaps without physical contact, i.e., without parasitic drag torque.

BASIC HINGE MECHANISM

Figure 1 shows the hinge mechanism's major components as designed for a development prototype. To clarify the basic components of the system, Fig. 2 presents a schematic of the mechanism. It should be noted that since this device was primarily intended for space use, all electric motors are provided with identical backups for use in case of primary motor failure.

The system is supported by two large self-aligning spherical roller bearings inserted in two pillow blocks. These roller bearings are mounted on stub shafts over conic sleeves which are forced under the inner races. This expands the inner races radially to eliminate all internal clearance and apply controlled radial preloads needed to meet launch requirements. As a result of the preloads, these bearings exhibit a significant drag torque which, together with the fluid coupler O-ring friction, provide the major contributions to the system parasitic drag torque. The two stub shafts are connected through a large cylindrical shell as shown in Fig. 1. Supported by flexure bearings, the stub shaft assembly can rotate through small angles with respect to the main hinge shaft. These flexures allow a total differential displacement of only $\pm 2°$ with a stiffness of 100-400 in-lb/rad depending on the thickness of the flexure blades.

The differential angular displacement between the stub shaft assembly and the hinge shaft is detected and measured by a sensor as shown in Fig. 2. This sensor controls the operation of a twist motor which adds sufficient torque to the stub shaft assembly to counteract all parasitic torque drag collected by the stub shafts. Thus, the primary hinge drive motor operates in a torque-free environment, thereby ensuring the desired hinge positioning accuracy.

The misalignment coupling in Fig. 2 is intended to provide enough compliance to accommodate manufacturing tolerances in shaft alignment at that point. This coupling allows for bending and for axial and lateral mismatching. However, its torsional compliance is not significant.

PARTIAL ANALYTICAL MODEL

In order to more clearly describe the operation of the powered hinge, it was found convenient to develop a simplified analytical model in which the masses, inertias, and details of the electronic loops were left out. This model is shown in Fig. 3.

It is shown in this system that a displacement provided by the drive motor acts directly upon the output branch and indirectly upon a parallel branch which, by means of a detector and a feedback control system, provides power to cancel out all parasitic drag torque.

An examination of the mechanism in Fig. 1 shows that some parasitic torque drag must necessarily exist in both branches of the system. However, by careful design, only one small ball bearing is left in the output branch to ensure proper alignment of the encoder wheel. In zero gravity, the torque drag of this bearing is expected to be at or near the noise level of the branch. All other sources of parasitic drag torque are collected on the other branch to be sensed by the drag torque eliminator system. These sources of parasitic torque include that of the main roller bearings, the smaller ball bearings, and any external torque drag (such as that of the rotating fluid couplings).

In operation, the position sensor detects an angular displacement $\Delta\theta$ and sends a signal to the twist motor. This signal closes the loop mechanically by rotating the stub shafts to cancel the displacement, thereby eliminating the drag torque.

The response characteristics of this control system are not addressed in this paper; however, the system is designed to function much faster than the primary drive system to ensure a very small lag angle $\Delta\theta$ and the intended drag-free operation of the powered hinge.

ACTIVE FLEXURE

The operation of the flexure is shown in Fig. 4, which illustrates one cycle of motion. The flexure consists of two concentric hollow shafts held together by thin flexible blades mounted as shown in Fig. 4-1. The two shafts have a small radial clearance such that large lift-off and landing radial and axial loads transmitted from the inner main shaft to the outer stub shaft can be taken by direct contact of the two shafts. The radial clearance between the two shafts is small enough that the flexure blades are not unduly stressed during static load. This clearance is also selected to ensure that no contact is made between shafts in the normal zero-g operation.

As can be seen in Fig. 4-1, the main shaft (output) is inside the stub shaft (drag eliminator system). The main shaft is connected to the main motor; the stub shaft is connected to the twist motor and runs with the drag-producing roller bearing.

When the main motor is activated, the main shaft rotates. Fig. 4-2 shows the small rotation θ , which brings the flexures almost in contact with the edges of the main shaft windows. The sensor, upon detecting this misalignment, energizes the twist motor, which rotates the stub shaft and restores a null position at the displacement angle θ , thereby driving the roller bearing and any other drag-producing components which may be connected to the stub shaft.

In a practical application, the angle θ is made as small as possible; in this instance, 1 arc sec The lag of the stub shaft over the main shaft is not perceptible.

EFFECT OF PARASITIC DRAG TORQUE ON SYSTEM RESPONSE

In order for this system to perform with the expected efficiency, the torque required to deflect the flexures must be as low as possible. This requirement implies a combination of low spring rate and small angular displacement. Bottoming out of the flexure is highly undesirable. Figure 5 shows a representation of a typical torque-vs.-displacement curve for a flexure where the bottoming angle is $\pm 2°$. In a well-designed system, the sensor must be capable of detecting a displacement of 1 arc sec. so that the flexures remain virtually undeflected with no parasitic torque being produced.

The residual torque along the output branch (see Fig. 3) must be very small, since it will not be eliminated.

All other sources of internal and external drag torque are applied to the "Eliminating" branch (Fig. 3) to avoid disturbing the output branch. The disturbance introduced by parasitic drag torque is shown in Fig. 6, which compares the displacement under the same torque of a one-degree-of-freedom system with and without friction drag. This plot shows how troublesome the effect of static friction is. In order to obtain a displacement, it is necessary to apply a torque at least equal to that produced by the static friction. However, as soon as the torque corresponding to the static friction is reached, the motion starts and the friction coefficient drops to the lower value corresponding to the dynamic condition. The system then jumps to a position of equilibrium, such as θ_1. By comparison, a friction-free linear system with the same stiffness and under the same torque would reach its equilibrium position at θ_2, going in a controlled manner through all intermediate angles as shown by the straight line passing through the origin.

Since friction is the major contributor to the parasitic torque drag, it is clear that a high-precision hinge cannot be designed unless it can be operated in a friction-free manner.

SOURCES OF PARASITIC DRAG TORQUE

In the general hinge system, sources of parasitic drag torque are to be found in the following components:

1. Main bearings (rollers or ball bearings)
2. Auxiliary motors bearings (ball bearings)
3. Encoder bearing (ball bearing)
4. Power line transfer system
5. Signals transfer system
6. Fluid lines transfer system

In the powered hinge discussed here, only Item 3, the encoder bearing, remains effective as a minor source of parasitic drag torque. Items 4 and 5 are transmitted by induction and capacitive coupling through r gaps (or vacuum gaps) while the remaining three, Items 1, 2, and 6, eliminated by the twist motor feedback loop.

CONCLUDING REMARKS

A full-size model of this powered hinge has been designed and built for test purposes. This model (Fig. 7) meets the requirements specified in this paper. It has been successfully subjected to qualitative running tests of the main drive motors without any drag relief.

Precision pointing tests have not been carried out at this stage because of delays in the design of the electronic feedback systems. However, analytical simulation techniques have shown that excellent controlled performance could be achieved at any selected angle. It should be noted that the hinge rotation angle is not limited, it can be a fraction or any number of revolutions.

In its present configuration, this precision powered hinge is a large device adapted to perform the functions represented by its specifications. Its large size was convenient for prototype fabrication, development, and testing. Application to smaller devices will require changes in its configuration to accommodate more severe space restrictions.

In addition to obvious aerospace applications, the principles of this mechanism should be adaptable in miniaturized form to robotics systems and other devices.

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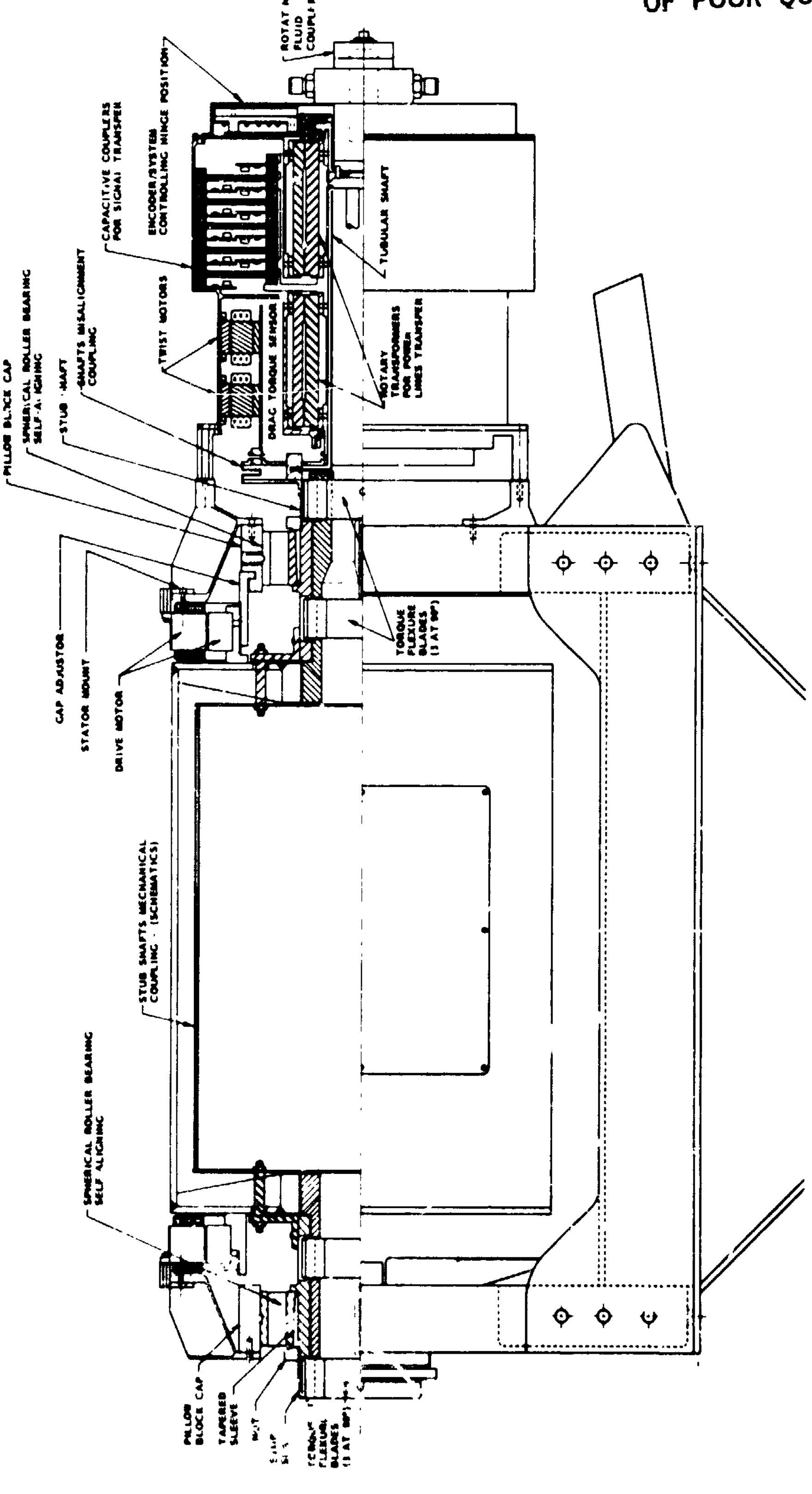

Fig. 1 Precision Hinge Mechanism

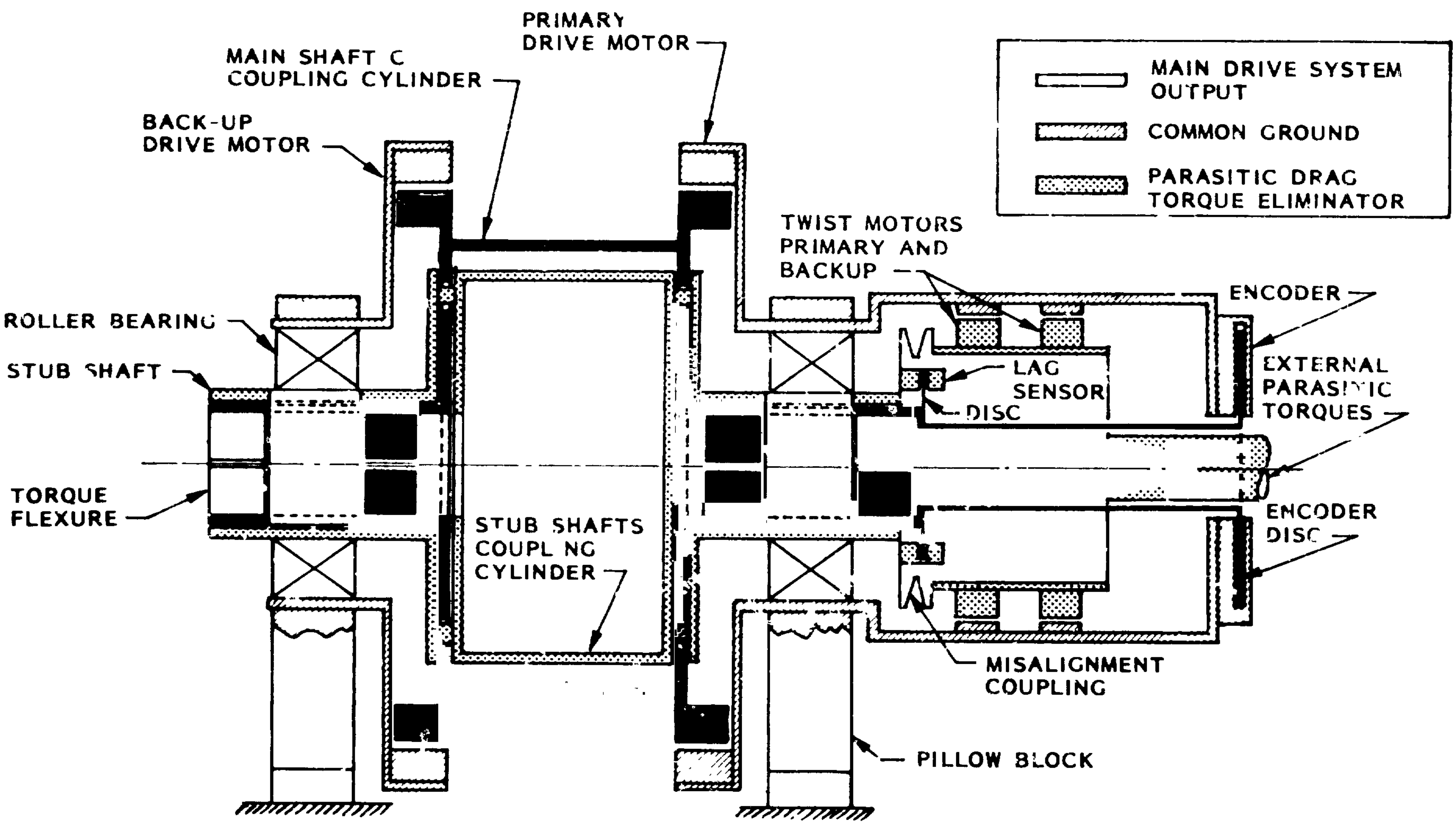

Fig. 2 Schematics of Basic Hinge Mechanism

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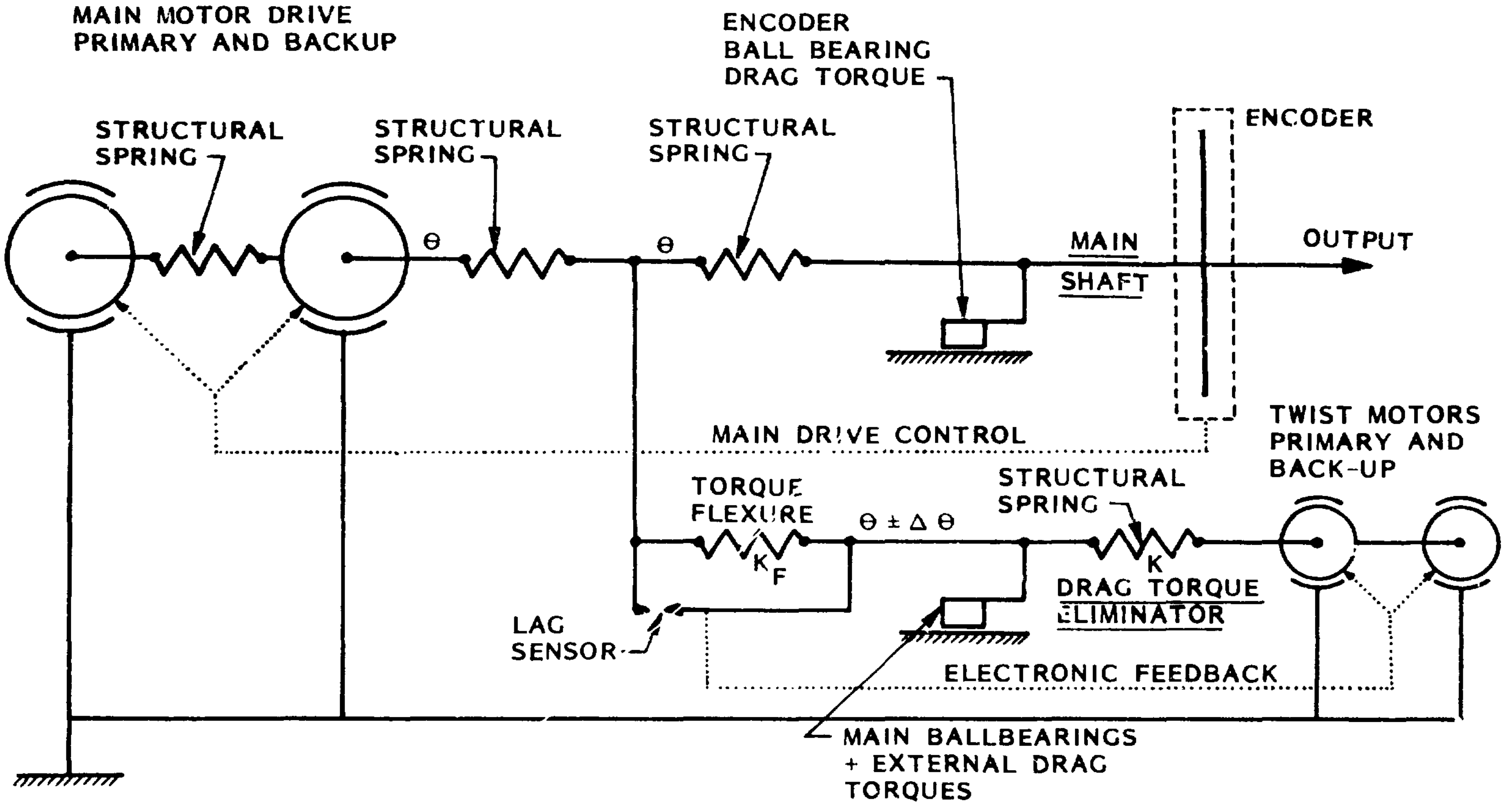

Fig. 3 Partial Analytical Model of Precision-Powered Hinge

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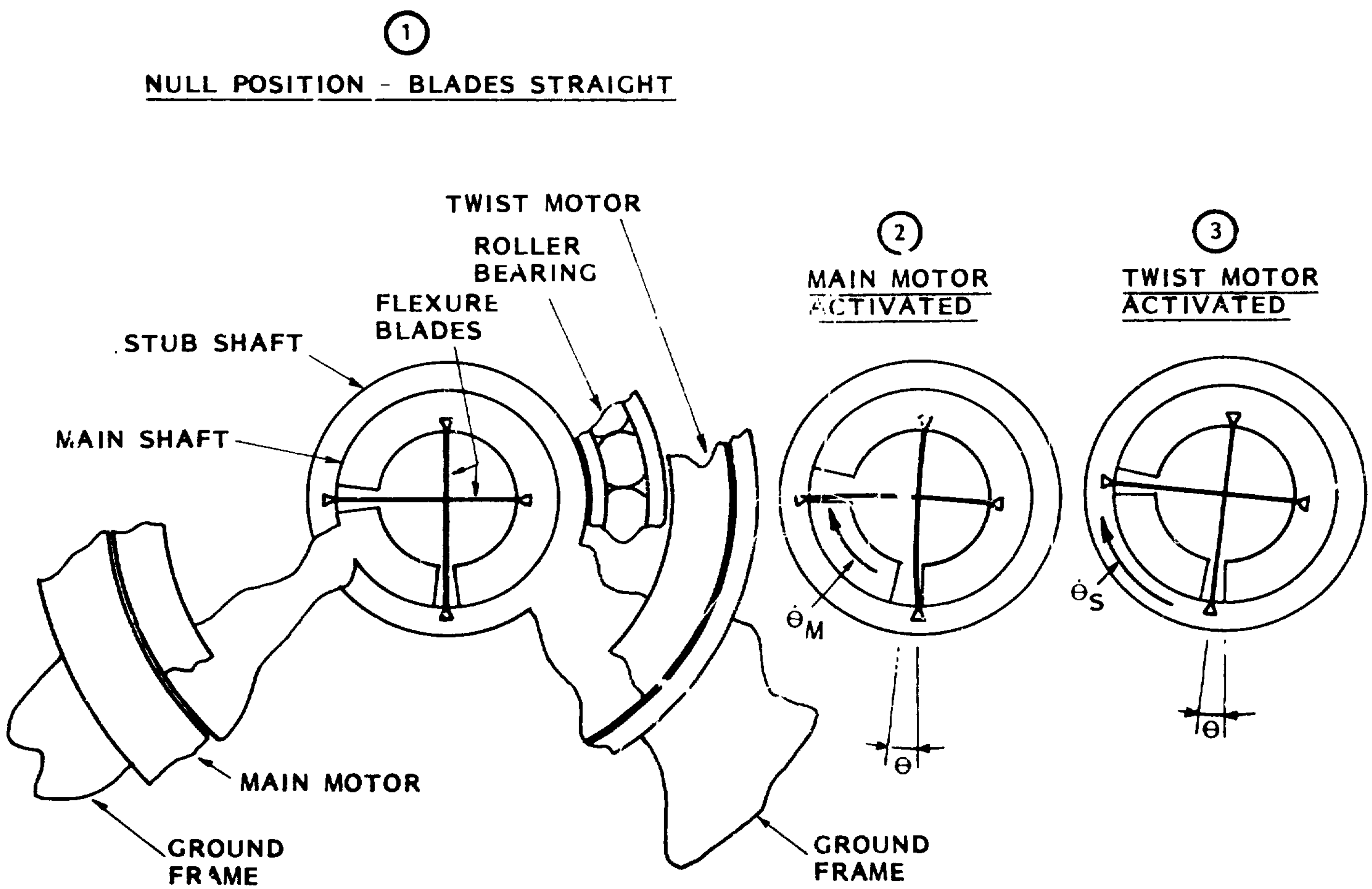

Fig. 4 Active Flexure Operation

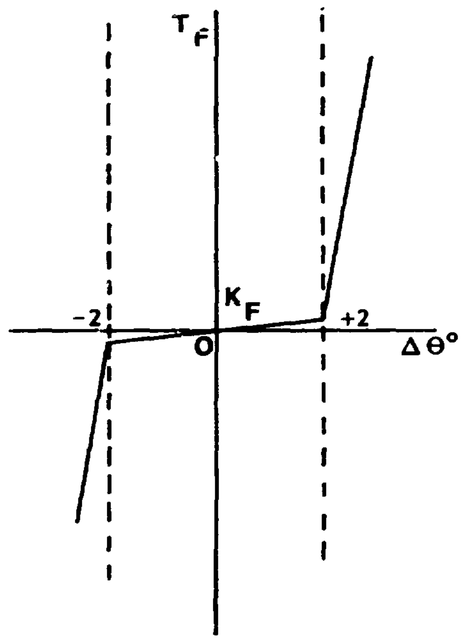

Fig. 5 Flexure Stiffness

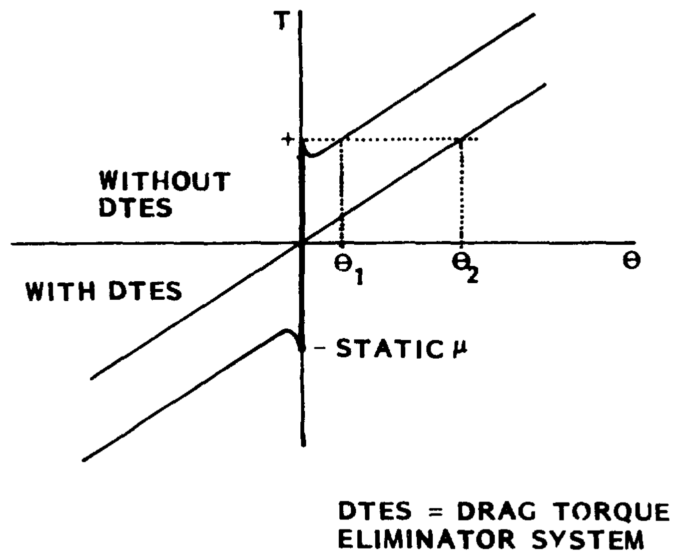

Fig. 6 Typical Torque-Displacement Curves With and Without Drag Eliminator System For a Simple One-Degree-of-Freedom Device

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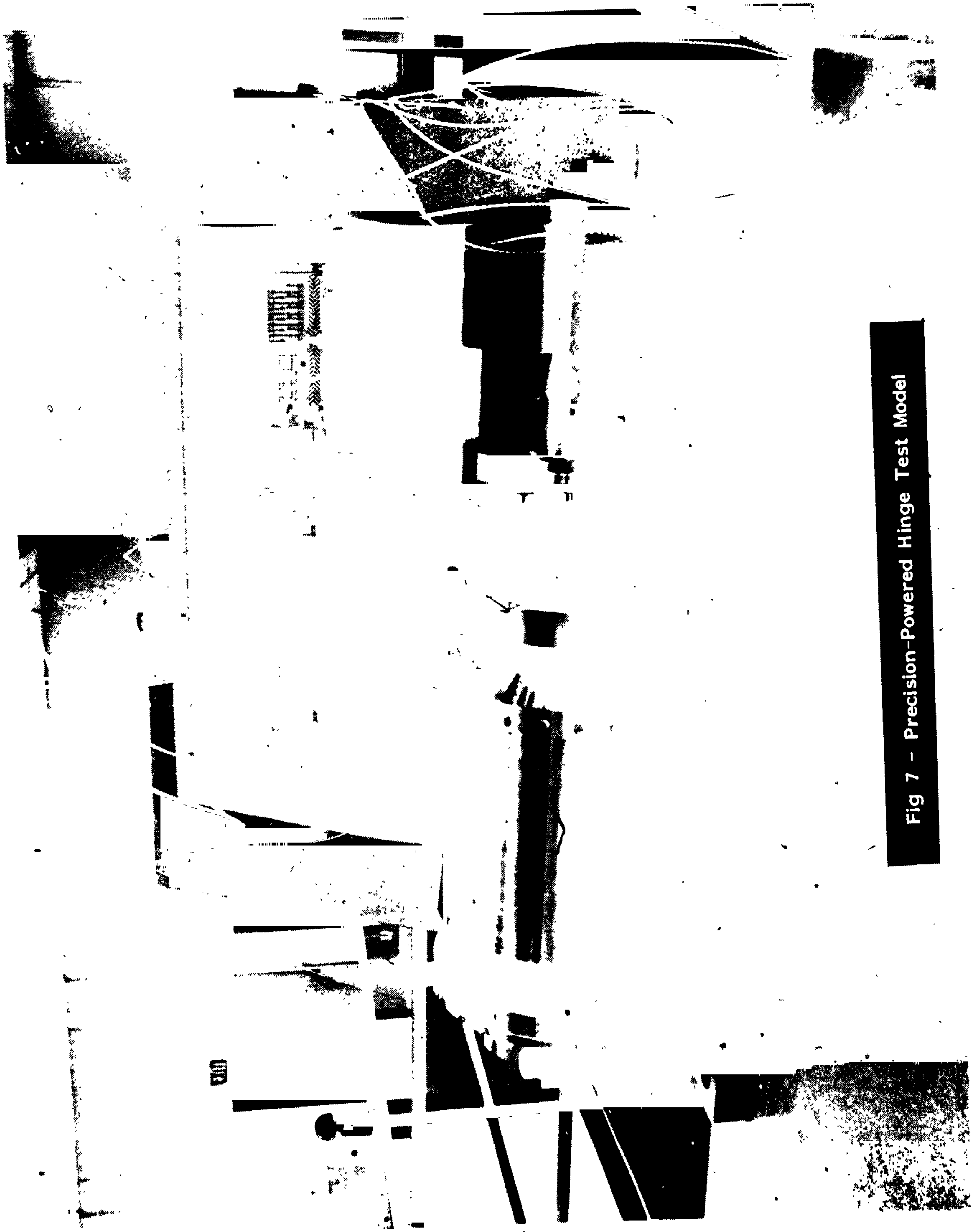

Fig 7 – Precision-Powered Hinge Test Model